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PASSIVE MOTION CONTROL OF PNEUMATICALLY DRIVEN DISPLACERS IN CRYOGENIC COOLERS

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The split cryogenic cooler with a remote cold finger offers many advantages for use in cooling of infra-red systems. A pneumatic drive for the displacer in such coolers has since been adopted in many cryocooler designs. This concept can be significantly improved by causing the displacer to move sinusoidally rather than in an essentially square wave, as in most of the present models. The way this motion has been achieved passively is described in detail and its advantages outlined. Data of minicoolers using the above mentioned concept is presented.

Key Words: Cryocooler; displacer; eddy current; magnetic damper; phase delay; split Stirling; Stirling cycle.

1. Introduction

One of the major advantages of the Stirling type cooler is its simplicity - being a valveless system. The gross cooling power Q_c of thermodynamic cycles in general and the Stirling cycle in particular, can be expressed by the equation:

$$Q_c = \oint_T PdV < 0 \; . \tag{1}$$

Note that negative Q_c means cooling. In steady state, both P and V are the periodical pressure and volume in the expansion chamber at a constant chamber temperature T. To a first approximation, both P and V very sinusoidally with time, and as shown directly from eq. (1), the volume change has to lag behind the pressure change to get cooling. For a given stroke and pressure pulse, the maximum gross cooling is obtained for a 90° phase delay of volume with respect to pressure. The optimal selection and the control of the phase angle between the pressure change and the expansion volume change has a great importance in determining the potential cooling power.

In the integral Stirling cooler the phase angle is maintained and kept constant by a crank mechanism. The Split Stirling version which gives significant advantage in some applications, for example to the electro-optic system designer, forces the designer to look for alternative solutions for the expansion volume cyclic change, and for the creation of the desired phase angle. A few mechanisms were developed and presented during the last decade; most of them are based on pneumatic drive of the expansion volume cyclic change and on a dry friction, viscous flow or electrical timing as a phase angle control mechanisms [1, 2, 3].

The major problem of the Split-type Stirling coolers is their relatively poor reliability. The primary factor affecting the reliability of this type of coolers is the dynamic seal wear in the compressor piston seal and the displacer seal of the expander unit. This statement is specially true in the case where the expander dynamic seal friction controls the motion of the displacer. Additional factors affecting reliability are solid or gaseous contaminants accumulated in the regenerator or around the displacer assembly, affecting its motion and degrading its efficiency. The "conventional" problems, such as external leaks, motor and bearing failure, can be considered a secondary mode of failure and are not part of this discussion.

2. Design Considerations

The basic approach of our new design concept is the elimination of the seal wear and contaminants modes of failure. The expander is based on a pneumatic drive. In order to allow high temperature vacuum baking, it is constructed of metal and of 3 Viton static O-rings only. The metal displacer has a screen type regenerator. The displacer and driving piston clearance type seals are constructed of metal (PH 5 - 15 Stainless Steel) and are located on the same plunger as shown in fig. 1. The all metal clearance type seals are characterized by their negligible wear rate and by their very low friction force.

The displacer motion phase delay control is governed by an independent passive mechanism located in the expander drive compartment. This mechanism consists of magnetic damper and helical suspension springs. By proper design of the damper mechanism, a near sinusoidal displacer motion - utilizing practically the full stroke available - with a prescribed phase delay relative to the pressure wave, can be achieved.

Assuming a sinusoidal pressure wave, the displacer driving force can be expressed by:

$$F_{drive} = A \, \Delta P \cos \omega t, \tag{2}$$

where A is the driving piston cross section area, ΔP = the difference between the maximum pressure and the mean pressure (equivalent to the pneumatic volume pressure) and ω is the angular frequency of the displacer. If the desired displacer motion is sinusoidal with, for example, a 90^{o} phase delay to the pressure wave, it can be expressed by the following quotation:

$$X = Xo \sin \omega t, \tag{3}$$

where Xo symbolizes half the maximum stroke possible. The sum of the forces affecting the displacer motion (neglecting the low value of the dry friction created by the clearance type seal) can be expressed by the following differential equation:

$$F_{drive} + M\ddot{X} + C\dot{X} + KX = 0 \ , \tag{4a}$$

$$A\Delta P \cos \omega t + M\omega^2 Xo \sin \omega t + CXo\omega \cos \omega t + KX \sin \omega t = 0 \ , \tag{4b}$$

where C is the damping coefficient, M is the sum of all the expander moving masses and K is the spring coefficient. As is shown clearly in these equations, the necessary conditions for balance are:

$$A\Delta P = CXo\omega \ , \tag{5}$$

$$M \omega^2 \approx K. \tag{6}$$

Equation 5 shows that the peak damping force must be equal to the peak driving force and eq. (6) shows that the inertial forces must be balanced by the proper selection of the springs. In addition, the springs determine the average position of the displacer stroke, which is important in order to obtain symmetrical displacer motion around midstroke. It is important to emphasize that the damping forces used in these equations are the sum of the drag created by the viscous flow in the regenerator and the force created by the additional damper. The generator viscous flow drag is a parasitic by-produce of the regenerator operation. The use of this drag as a sole damping mechanism to control the proper phase angle (as applied in one of the alternative designs) (2), enforces a compromise in the regenerator design between the thermodynamic and the viscous drag requirements.

The magnetic damper, as used in our design, is shown in fig. 2. This device is basically a miniature eddy current generator in which an inverted cup-shaped copper body is reciprocated in radial magnetic field - created by rare earth permanent magnet rings. The copper cup shape is

bolted to the end of the displacer driving piston and therefore, both move simultaneously. The eddy currents generated by the copper cup moving in the magnetic field create a mechanical drag force proportional to the displacer linear velocity. Proper selection of the magnetic damper design parameters, taking into consideration the regenerator viscous flow drag, will result in smooth full stroke sinusoidal displacer motion and in the desired phase angle. The amount of damping forces related to the linear velocity and to the operational frequency, can affect the displacer stroke as shown in fig. 3. The damper design point must be selected to balance the driving force with the expander at its cold, steady state, condition.

By using the dry friction phase delay mechanism, almost rectangular in shape, indicator diagrams can be achieved. This results in maximum cooling power achievable in the given pressure and volume change made. The sinusoidal displacer motion, however, results in elliptical shape diagrams with an area about 20% smaller. This reduction is compensated by the lower dynamic thermal losses which depend on the working gas mass flow rate. These rates are much higher in the square type diagram than in the smooth elliptical shape diagram. The smooth non-contacting sinusoidal displacer motion is a very desirable characteristic from the acoustical and mechanical noise level point of view.

3. Conclusion

1/4 W Split Stirling coolers based on the eddy current damper and clearance seal design concepts are in production by RICOR LTD. in Israel since 1982. Several hundreds of coolers have been delivered and are accumulating laboratory and field experience. A 1/4 W Split Stirling cooler of this concept is shown in fig. 4. Several reliability demonstration tests of the expander units were performed as part of qualification programs and a proven MTBF higher than 1500 hours was demonstrated. Actually, the expander reliability is limited by contamination originated in the compressor, and accumulating on the cold surfaces inside the regenerator, around the displacer tube and in the clearance seals. The combination of our next generation contamination-free compressors and the present magnetic damper/clearance seal type expanders, will result in reliability greater than 2500 hours.

4. References

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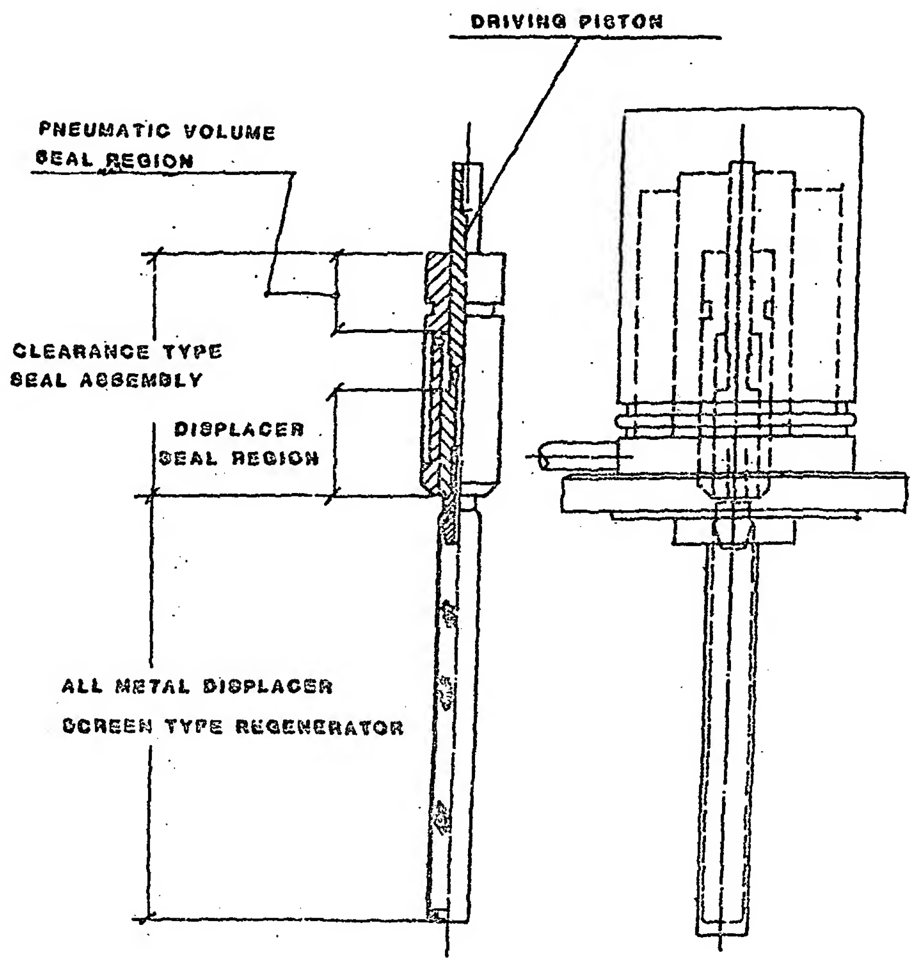

Figure 1. Expander clearance type dynamic seals.

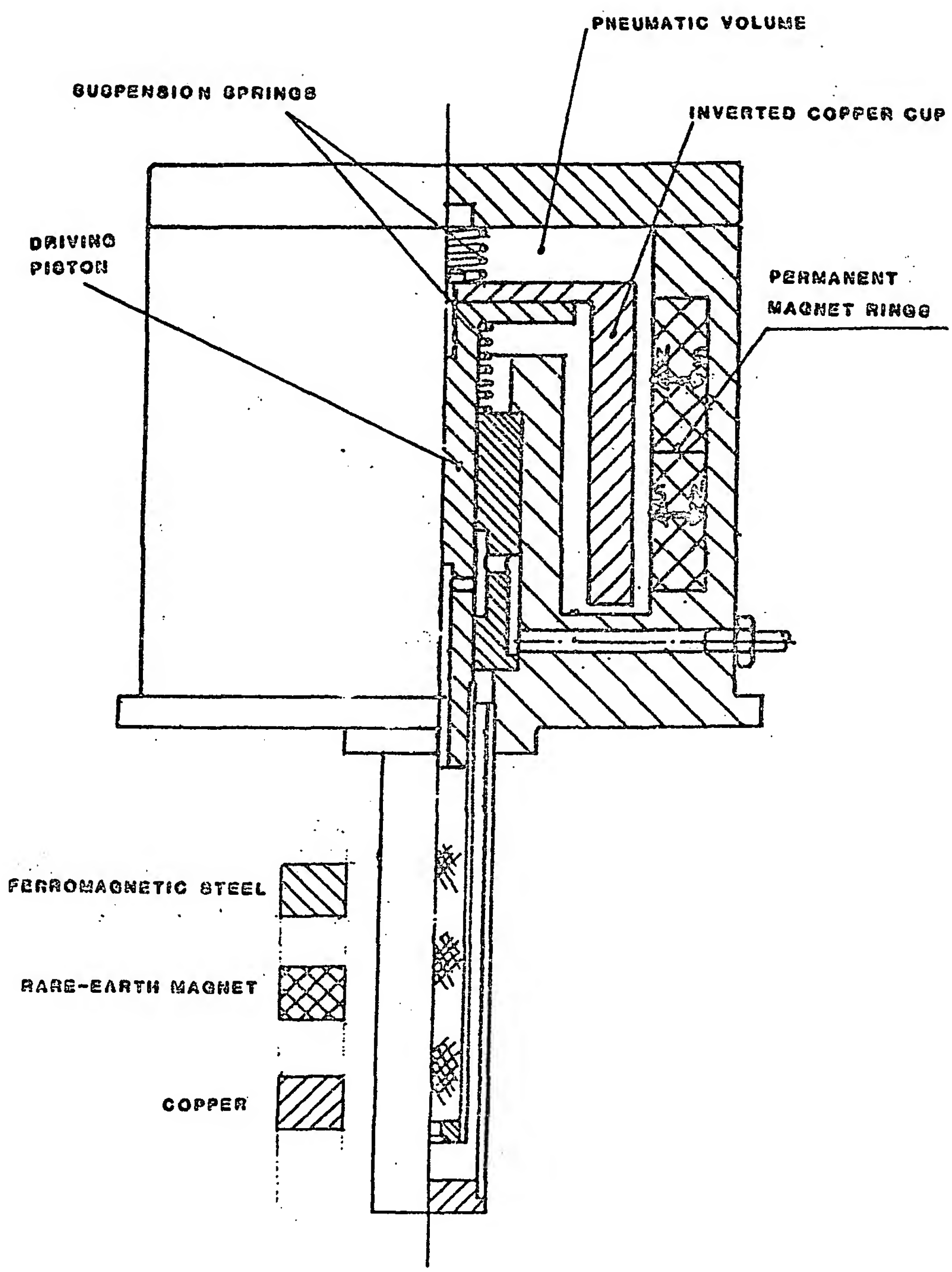

Figure 2. Magnetic damper - Conceptional scheme.

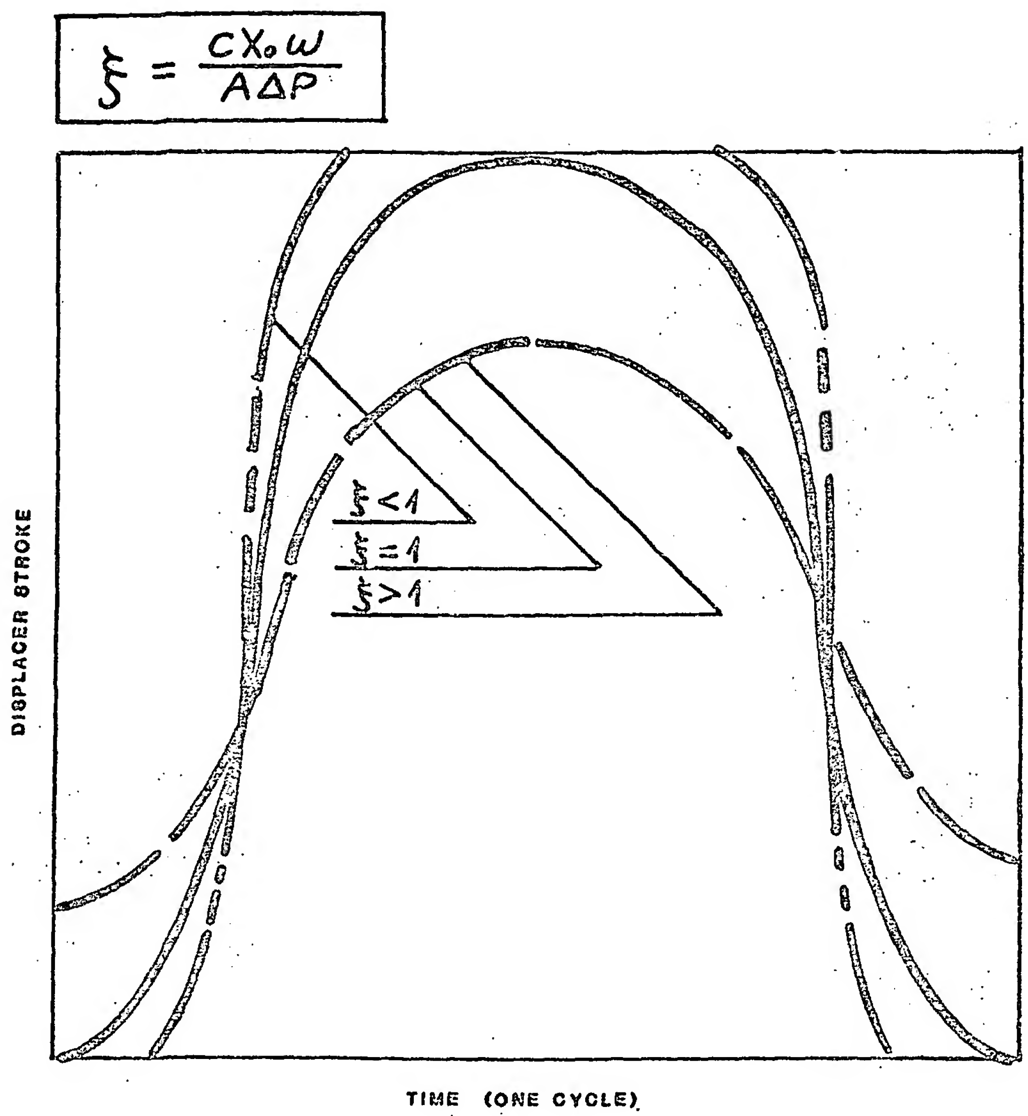

Figure 3. Displacer motion at different damping levels at constant frequency.

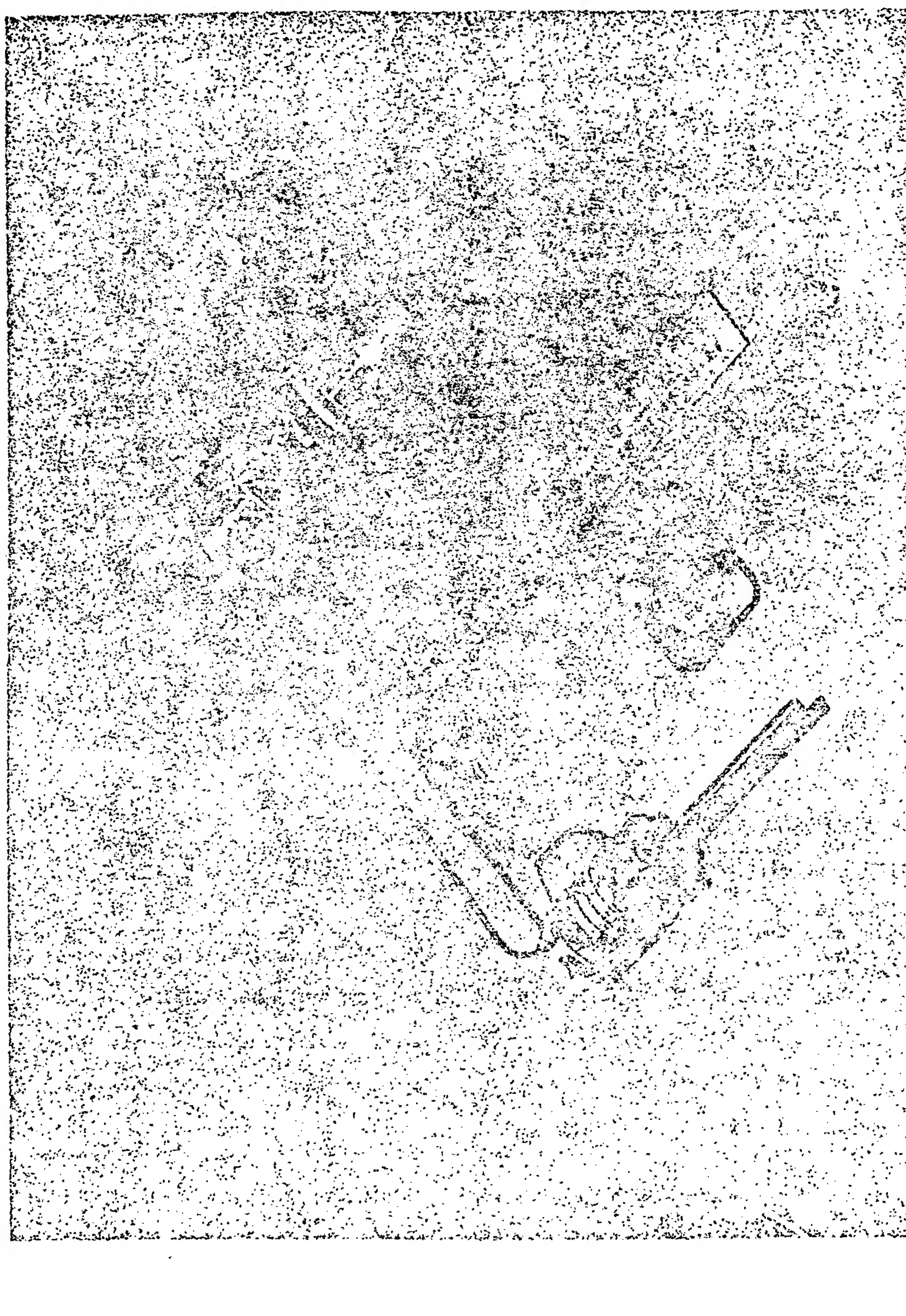

Figure 4. Ricor Mode K413H 1/4 split Stirling cooler - general view.